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ABSTRACT

As well known, the performance of the compressor is concerned with the corresponding refrigeration system. In this paper, by comparing three types of refrigeration cycle, an optimized range of the intermediate pressure for gaining better coefficient of performance (COP) is proposed. Additionally, the effect of the cycle on the construction of the compressor and performances under different refrigerants are discussed. The simulation result indicates that the coefficient of intermediate pressure under freezing condition and refrigeration condition ranges from 0.81 to 0.91 and 0.93 to 0.97, respectively. The results obtained here may provide some guides for the optimal design and operation of practical refrigeration system.

1. INTRODUCTION

In recent years, for the purpose of energy-saving, high efficiency compressor has received great attentions. Many researchers have contributed a lot in the view of compressor structure optimization. But since the performance of compressor is related to the corresponding refrigeration system, some researchers have proposed several refrigeration cycle to optimize the performance, especially in the heat pump system under low ambient temperature conditions. Bertsch (2008) compared many different heat pump cycles and concluded that cascade cycle and two-stage cycle outperform the conventional cycle. Heo et al. (2011) compared the heating performances of air-source heat pump cycle with different types of refrigerant injection, revealing that the vapor injection would enhance heating capacity significantly. Wang et al. (2015) compared the basic flash tank vapor injection with a novel ejector enhanced vapor injection cycle, pointing out the prominent advantages in enhancing the performance and reducing the compressor discharge gas temperature. But as we all known, when applying cycles to business, especially in the household air-conditioning market, the complexity of the system is required to be reduced. In this paper, three types of simple refrigeration system are proposed, and simulated with different refrigerants under ASHARE refrigeration and freezing standard operating conditions. By analyzing the intermediate pressure factor, optimal range for obtaining maximum COP and greater heating capacity is proposed. The results obtained here may provide some guides for the optimal design and operation of practical refrigeration system.

2. MODELING

Figure 1 show three types of the refrigeration system, which are different from those mentioned about in the introduction part. In other words, they are much easier to apply to household air conditioners due to their minor changes. What's more, their schematic system and corresponding p–h diagrams are shown in Figure 2, 3 and 4. As well known, all the refrigeration systems operate basing on the Basic-Cycle (BC). Refrigerant flows into the compressor to obtain high pressure, and becomes low pressure by flowing through the condenser and expansion valve. Different with Basic-Cycle, the Jet-Cycle (JC) separates the expansion valve outlet refrigerant into two flows by gas-liquid separator. The saturated steam flows into the mixing room of the compressor, while the saturated liquid is throttled again and flows through the evaporator and the first-stage cylinder of the compressor. In the compressor, two strands of fluid mix together in the mixing room, and then compressed in the second-stage cylinder. The flow paths are shown in Figure 3.

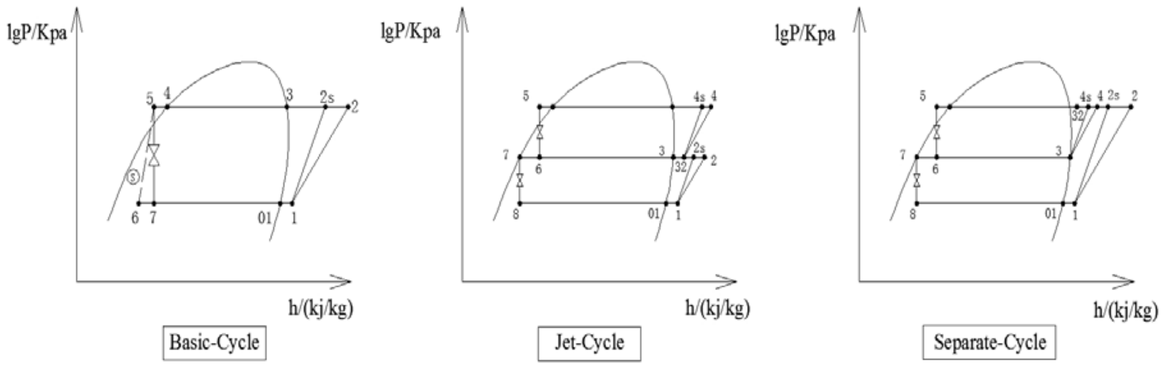


Figure 1: P–h diagrams comparison

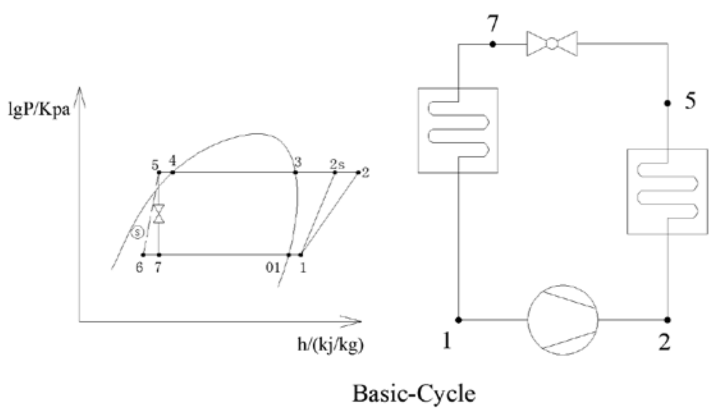


Figure 2: BC schematic system

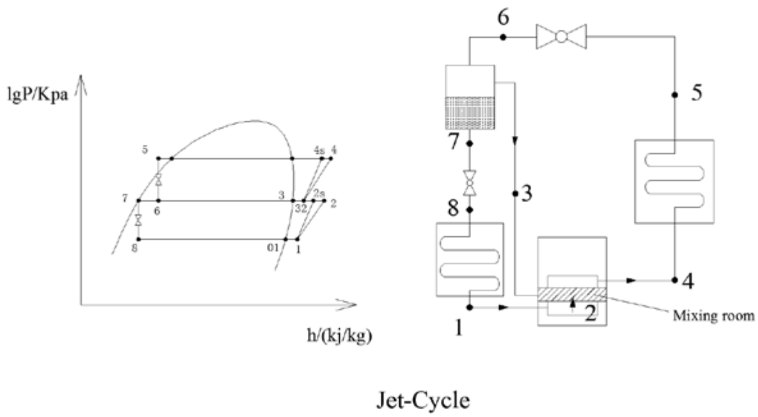
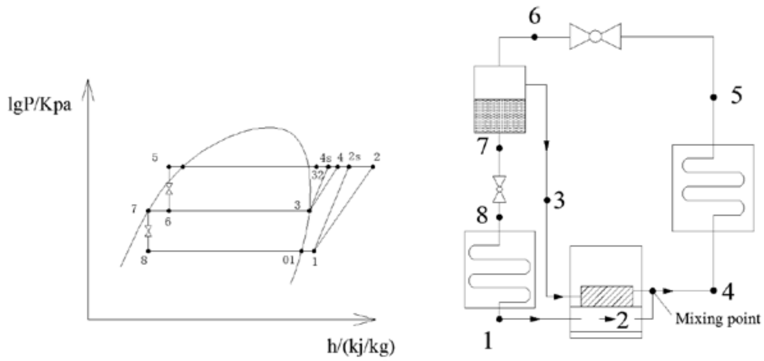


Figure 3: JC schematic system



Separate-Cycle

Figure 4: SC schematic system

As described above, two fluids mix together in the compressor mixing room in the JC. But in Separate-Cycle (SC), which flow paths are shown in Figure 4, two fluids mix together in the compressor outlet but not the mixing room. Each flow compressed in separated cylinder. Actually, in the air-conditioning compressors, high pressure refrigerant gas is exhausted into the shell. Although JC and SC have been proposed theoretically many years ago, they have only been applied to household air conditioner recently. The purpose of this paper is to analyze and compare the system performances in order to obtain optimal intermediate pressure. The cooling capacity, heat capacity, coefficient of performance (COP) and the intermediate pressure λ factor are calculated as

$$Q_c = q_e \times (h_1 - h_8) \text{ (JC, SC)} \quad Q_c = q \times (h_1 - h_7) \text{ (BC)} \tag{1}$$

$$\begin{aligned} Q_h &= q \times (h_4 - h_5) \text{ (JC)} \\ Q_h &= q \times (h_{32} - h_5) \text{ (SC)} \\ Q_h &= q \times (h_2 - h_5) \text{ (BC)} \end{aligned} \tag{2}$$

$$P_m = \lambda \times \sqrt{P_e P_c} \text{ (JC, SC)} \tag{3}$$

$$\begin{aligned} W &= q_e \times (h_2 - h_1) + q \times (h_4 - h_{32}) \text{ (JC)} \\ W &= q_e \times (h_2 - h_1) + q_m \times (h_4 - h_3) \text{ (SC)} \end{aligned} \tag{4}$$

$$COP_h = Q_h / W \text{ (BC, JC, SC)} \tag{5}$$

where q_e, q, q_m are the mass flow of the evaporator, condenser and intermediate-stage respectively, P_m, P_e, P_c refers to the intermediate, evaporating and condensing pressure and λ is regarded as an intermediate pressure factor.

3. RESULTS AND DISCUSSIONS

Figure 5 shows the cooling capacity and heating capacity of different systems. In this paper, the temperature are selected according to the ASHARE refrigeration (54.4/7.2 ℃) standard operating conditions or ASHARE freezing (54.4/-23.3℃) standard operating conditions when the refrigerant is set to be R410a, R32, R290 and R600a. It can be easily seen from Fig.5 that with the increase of the intermediate pressure factor when the refrigerant is R410a, the cooling capacity and heating capacity increases both. But when the λ_{410} is lower than 1.0, the increment speed of the cooling capacity is greater than the heating capacity, which means there exits an optimum coefficient of performance (COP) of JC and SC both. In addition, conclusions can be easily obtained that the performance of JC and SC are almost the same, and when the λ_{410} is greater than 1.5, both cooling capacity and heating capacity of the JC and SC will larger than that of BC.

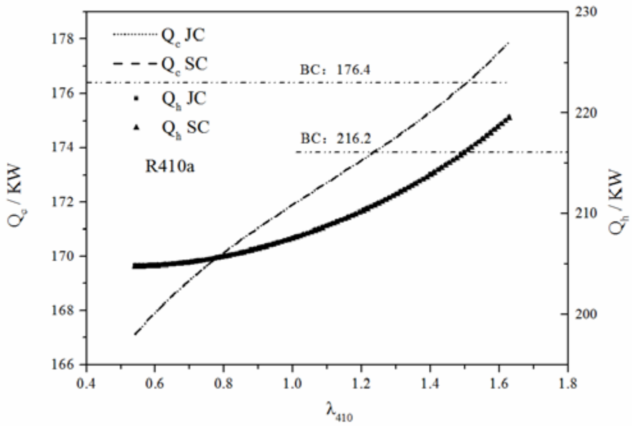


Figure 5: Variation of cooling and heating capacity with different intermediate pressure factor

Figure 6 shows the variation of COP of both JC and SC with different intermediate pressure factor λ_{410}. The inference of fig.5 that there exits an optimal COP is confirmed. With the increase of the λ_{410}, the COP of both JC and SC increase first, and reach the maximum value at $\lambda_{410}=1$ approximately. In addition, it is obvious that in a large range, the COP of BC is lower than both JC and SC. From fig.5 and fig.6, conclusion can be easily obtained that in the case of obtaining better cooling capacity or heating capacity, the intermediate pressure factor λ_{410} of JC and SC should be selected to be larger than 1.5, but in the case of obtaining the optimal COP, λ_{410} of JC and SC should be selected to be 0.95. Although the maximum COP of SC is larger than that of JC in Fig.6, considering the calculation accuracy error, the performance of the JC is regarded as identical as SC. But in fact, in the actual compression process, refrigerant of JC flows through two vents, leading to a larger flow resistance compared with SC, which only has one exhaust vent. In other words, the performance of the SC is better than the JC actually, even though they are identical theoretically.

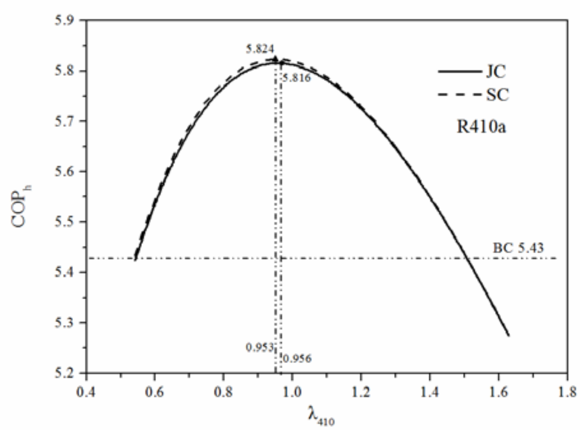
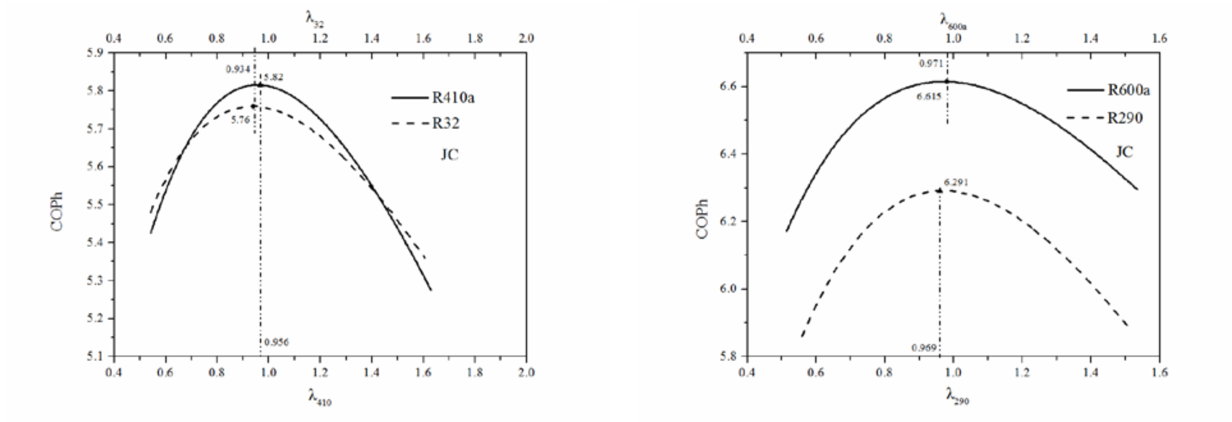


Figure 6: Variation of coefficient of performance (COP) with different intermediate pressure factor

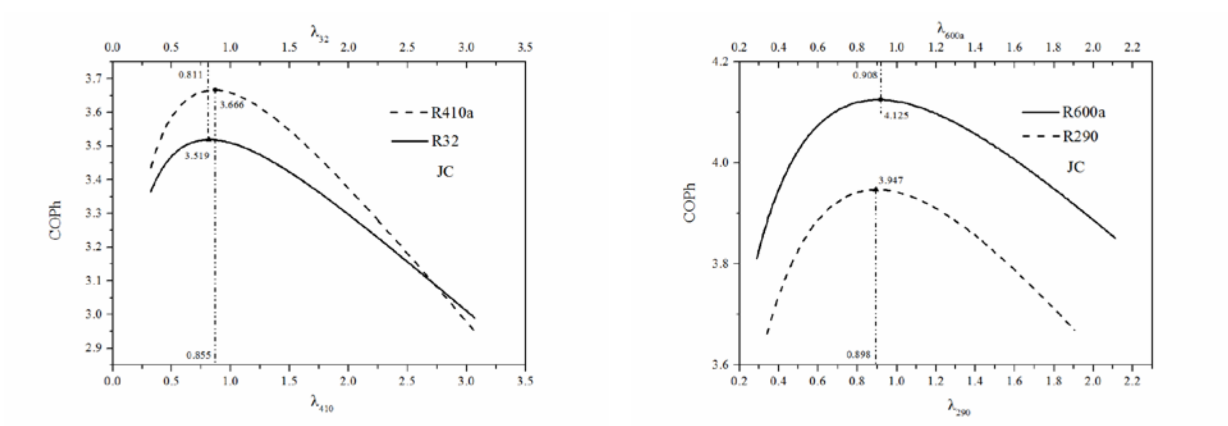
To some extent, the refrigerant type and the operating conditions can both affect the intermediate pressure factor when obtaining the optimal COP. In order to better analyze the influence of refrigerant and operating condition on the intermediate pressure factor, investigation are conducted in this paper, and corresponding results are demonstrated in Fig. 7 and Fig 8. In fig.7 (a), the variation of COP of SC with λ

under different refrigerant R410a and R32 is shown, and fig.7 (b) is under R290 and R600a. Figure 8 is the situation as same as fig.7 under freezing operating conditions. Specific data are shown in Table 1.



(a) (b)

Figure 7: Variation of coefficient of performance (COP) with different λ under ASHARE refrigeration condition when refrigerant is R410a and R32 (a), R290 and R600a (b).



(a) (b)

Figure 8: Variation of coefficient of performance (COP) with different λ under ASHARE freezing condition when refrigerant is R410a and R32 (a), R290 and R600a (b).

Table 1: Specific data of different conditions of JC

Condition	**Refrigerant**	**COP_{hmax}**	**λ**
ASHARE refrigeration	R410a	5.816	0.956
	R32	5.759	0.934
	R290	6.291	0.969
	R600a	6.615	0.971
ASHARE freezing	R410a	3.666	0.855
	R32	3.519	0.811
	R290	3.947	0.898
	R600a	4.125	0.908

By analyzing data of Table 1, conclusion can be easily obtained that the intermediate pressure factor λ ranges from 0.81 to 0.91 and 0.93 to 0.97 under the ASHARE freezing and ASHARE refrigeration operating condition, respectively.

4. CONCLUSIONS

In this paper, simulation analysis on three types of refrigeration system under different operating conditions is presented. An optimized range of the intermediate pressure for gaining better coefficient of performance (COP) is proposed. Additionally, the effect of the cycle on the construction of the compressor and performances under different refrigerants are discussed. The simulation results are summarized as follows:

- In JC and SC, both the cooling capacity and heating capacity increase with increase of intermediate pressure factor λ, but there exits an optimal coefficient of performance (COP).
- By selecting a suitable λ, both the cooling, heating capacity and COP of SC or JC can exceed that of BC.
- The performance of the SC is better than the JC actually, even though they are identical theoretically.
- Optimal λ for obtaining maximum COP ranges from 0.81 to 0.91 and 0.93 to 0.97 under the ASHARE freezing and ASHARE refrigeration operating condition, respectively.

NOMENCLATURE

COP	Coefficient of performance	
h	Enthalpy	(KJ/Kg)
q	Mass flow	(Kg/s)
Q	Heat	(W)
W	Power consumption	(W)
λ	Intermediate pressure factor	

Subscript

e	evaporator
m	intermediate
c	cooling
h	heating

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